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Water as a thermoacoustic working fluid

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This short report, addressed only to the thermoacoustic cognoscenti, discusses thermodynamic and transport properties of water with emphasis on water's virtues as a thermoacoustic working fluid. Short-stack-approximation calculations are presented, showing that water is a good working fluid. A very rough design for a sound source using water is also presented, as a starting point for discussing the merits and difficulties of this technology.

1. Motivation

To date, we have used air, helium gas, and liquid sodium as working fluids in our thermoacoustic heat engines. The present study of water as a working fluid was begun because water is the most common, safest, least expensive fluid on our planet, and hence the possible simplicity of using water in engines is appealing. This simplicity is most attractive in the ocean, as illustrated here by two examples of possible applications.

First, it is desirable to use very low frequencies in underwater acoustics because low-frequency sound suffers less attenuation than high-frequency sound. However, low-frequency sound sources using conventional piezoelectric technology are complex and inefficient: Shipboard heat is converted to electric power, which is then converted first to dc and then to ac electric power of the desired frequency, which then drives piezoelectric materials arranged in a "flextensional" or other leveraged arrangement to produce the large displacement amplitudes required at low frequencies. The possibility of converting heat directly to acoustic power is most appealing, especially if water itself can be used as the working substance. Such a sound source might be useful for continental-shelf petroleum exploration.

A second possible application of ocean thermoacoustics is the use of heat from deep-sea hydrothermal vents to do something useful such as generate electricity. This possibility was first suggested to us by Charles Stuart of

DARPA. Enormous quantities of heat are delivered by these vents at temperatures as high as 650^{o}K, but any equipment designed to use this heat would have to be exceptionally simple and reliable because of the great difficulty of installing, repairing, and maintaining anything at such depths. Interestingly, such high-pressure water was used as a heat engine working substance in the 1920's by J. F. J. Malone, who built a 50-horsepower coal-burning Stirling engine that used 700-bar water as its working fluid!

2. Relevant properties of water

The two most important thermophysical properties of a thermoacoustic working fluid are the thermal expansion coefficient $\beta = -(1/\rho)\ (\partial\rho/\partial T)_p$ (where ρ is density, T is temperature, and P is pressure) and the Prandtl number $\sigma = \mu C_p/K$ (where μ is dynamic viscosity, C_p is isobaric specific heat, and K is thermal conductivity).

The thermal expansion coefficient must obviously be the most important property a heat-engine working fluid has, as it couples heat flow to mechanical power. For the ideal gas, the most common heat-engine working fluid, we all know that $T\beta = 1$. Most of us have the seriously flawed idea of $T\beta$ for an arbitrary fluid shown in Fig. 1(a). We naively expect that vapors are essentially ideal gases, with $T\beta = 1$, while liquids essentially have a density independent of temperature (and pressure), so that $T\beta \simeq 0$. We may know that most thermodynamic derivatives go to infinity at the critical point, and we're probably not sure what happens elsewhere.

In Fig. 1(b) we show contours of constant $T\beta$ in the P-T plane for water. These curves are very similar for other fluids, when plotted on axes normalized by the critical temperature and pressure. We see that water vapor does indeed behave much like an ideal gas. Liquid water is, however, far more expansive, more powerful, than we would naively expect. At room temperature, $T\beta < 0.1$; but everywhere above 150^{o}C, liquid water has a large enough expansion coefficient to be an excellent heat-engine working fluid. And $T\beta > 1$ occurs over a very large region, not just in the immediate vicinity of the critical point.

We mention a few facts to put the axes of Fig. 1(b) in context for practical ocean applications. We note that common structural materials, such as stainless steel, lose considerable strength above 600-700^{o}C, so that simple devices will have to operate below 700°C. Judging from hydrophone specifications in EDO Corporation's catalog, we guess that underwater acoustic exploration operations are generally carried out at depths such that the pressure is less than or equal to 50 bar. The deep-sea hydrothermal vents are at far greater depths, where the pressure is greater than 300 bar.

The other thermophysical quantity of key importance in thermoacoustics is the Prandtl number σ, basically a dimensionless ratio of viscosity to thermal conductivity. Thermoacoustic engines work best for low-Prandtl-number fluids. This is because considerable heat-transfer surface area is required in the "stack" of a thermoacoustic engine, and viscous shear occurs on all that surface area, dissipating acoustic power. For the monatomic ideal gas, $\sigma = 2/3$, a value that we know is acceptably low for practical applications. Figure 2 displays σ for water. It is somewhat greater than for monatomic gases; but only below 100°C, where $T\beta$ is prohibitively small anyway, is σ really large.

3. Rough estimates for short engine

To make a quick, semi-quantitative survey of water's suitability as a thermoacoustic working fluid, we used the short-engine approximation [Eqs. (76) and (80) in the review/tutorial article, J. Acoust. Soc. Am. **84**, 1145 (1988)] to compute power density and efficiency at various values of T and P. Thermophysical properties of water are computed with a code written by Andrew Fusco, based on the 1967 Steam Tables. Fresh water and salt water are essentially identical for these purposes. A large number of assumptions were built into the FORTRAN code for this short-engine thermoacoustic calculation, so that the results would be simple enough to interpret at a glance. These assumptions are reasonable, based on our other experience in thermoacoustics; but we will not be surprised if experiments and more sophisticated calculations eventually show factor-of-two disagreement with these short engine results.

The assumptions built into these calculations are: 1) The plate spacing is four thermal penetration depths. 2) The plates are stainless steel and their thickness is chosen so that $\epsilon_s = 0.1$, if possible subject to the constraint that the plate thickness be greater than 0.005 inches and less than the plate spacing. 3) The acoustic pressure amplitude P_A is as large as possible, subject to four constraints: a) the total energy flow $\dot{H}_2 < 300\ W/cm^2$ so that effective heat exchangers are possible; b) the Reynolds number based on viscous penetration depth is less than 10^3, so that the flow is laminar; c) $P_A < 0.8|P-P_{sat}|$, so that the acoustic oscillations do not carry the fluid to the liquid-vapor equilibrium line; d) $P_A < 0.1\ P$ both in the vapor and for $T > T_{crit}$, so that nonlinearities arising from the $(\vec{v} \cdot \vec{\nabla}) \vec{v}$ term in the Navier-Stokes equation can be neglected. 4) The stack position and stack length are chosen to maximize $[\eta/\eta_c]^3[\dot{W}/A\Delta T]$ where η/η_c is the efficiency normalized by Carnot's efficiency and $\dot{W}/A\Delta T$ is the acoustic power generated per unit area of resonator per unit temperature difference across the stack.

The results of these calculations are shown in Fig. 3, for a frequency of 100 Hz. The efficiency is high, about 15-20%, everywhere except at very low temperatures where both β and σ are unfavorable, and in small regions above but near the critical point where, apparently, the deleterious effects of large σ overwhelm the beneficial effects of large β. (This effect deserves more attention some day.) The power density is generally in the range of 10^{-1} - 10^{-2} W/cm^2-K, and is generally higher at higher P where higher P_A is possible. Tom Gabrielson at NADC/Warminster has independently calculated efficiency and power density in a more limited regime, in agreement with these results. Overall, water looks like a perfectly satisfactory working fluid. Results at 500 Hz are similar to those at 100 Hz: Water is satisfactory at essentially all frequencies.

4. What a large underwater sound source might look like

We conclude this report with a brief, crude design of a heat-driven sound source suitable for use in the ocean at a depth of 1000 feet. The device is shown in Fig. 4. We don't know if this device satisfies any realistic needs; we only present it as a starting point for discussing the qualitative merits and difficulties of this technology. We leave many questions unposed!

We use the vapor phase at a pressure of 30 bar, with a pressure amplitude of 3 bar at the standing-wave antinodes. The resonator is closed at the top and open at the bottom, so the liquid-vapor interface at 240°C at the bottom of the resonator is gravitationally stable. Because of the large impedance mismatch between liquid and vapor, the resonator's fundamental mode has a half wavelength filling the tube length; at $f = 200$ Hz, $\lambda/2 = 1.3$ m.

We suppose the engine operates between 700°C and 250°C. Figure 3 shows $\dot{W}/A\Delta T = 0.01$ W/cm^2-K, so $\dot{W} = 9$ kW of acoustic power is produced in the 50-cm-diameter resonator. Radiation of this power into the open water surrounding the device produces 210 dB re 1 μPa @ 1 m. It will be necessary to reduce the diameter of the bottom opening of the resonator to about 25 cm to radiate this amount of power into the liquid water. A larger diameter opening would radiate too much power, draining too much energy out of the resonance and killing the oscillations. A smaller diameter opening would trap too much power in the resonator, so that some of the 9 kW produced in the stack would increase the amplitude of the resonance, until nonlinear processes in the resonator began dissipating power.

Figure 3 also shows that $\eta/\eta_c \simeq 18\%$, so $\eta = 8\%$ for these temperatures. Thus 100 kW of heat are required to drive the engine. Chemical heat sources such as Li-SF_6 have an energy density of the order of 10^4 J/gm, so that the device consumes 10 gm/s $\simeq$ 100 lb/hr of fuel.

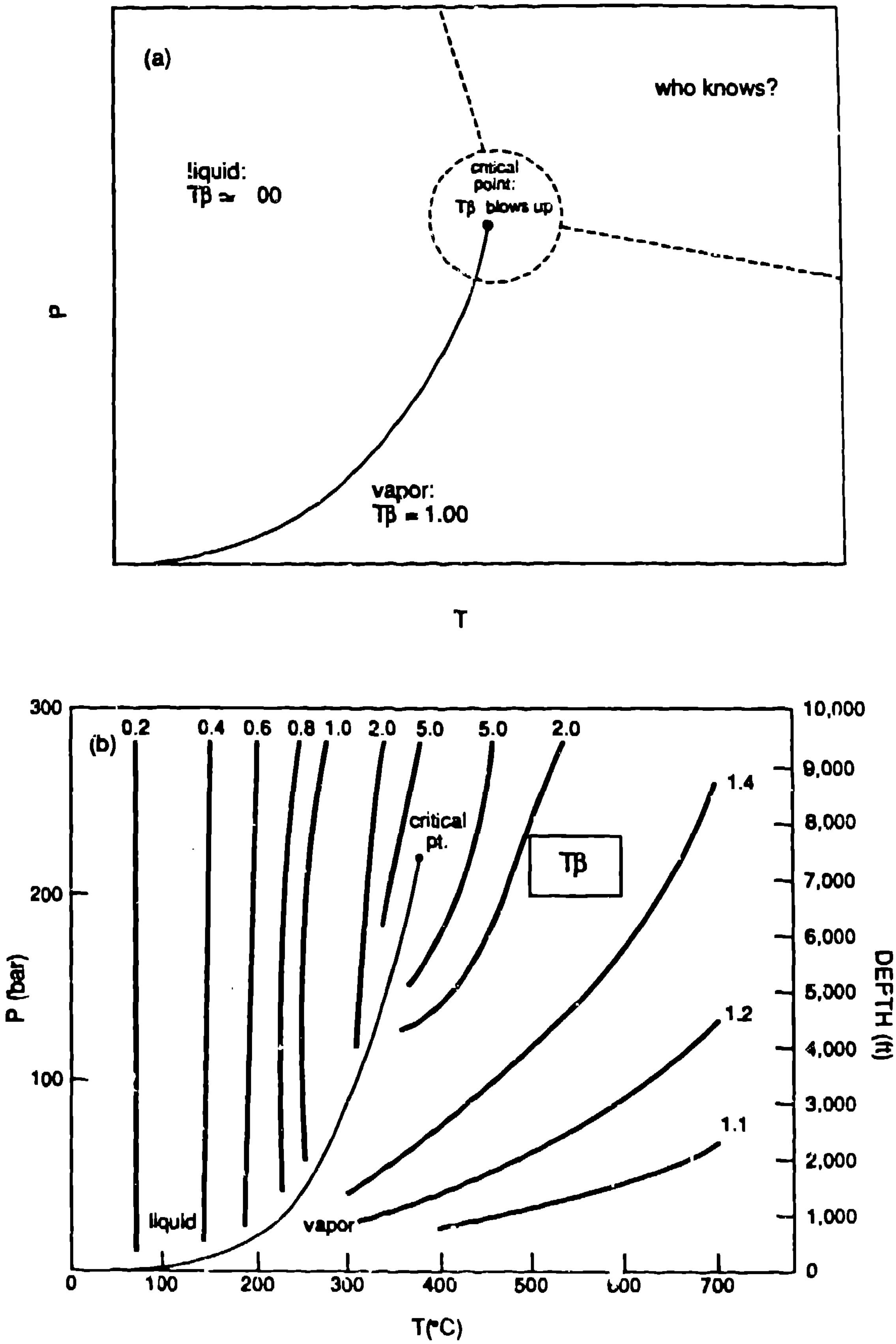

Fig. 1 (a) Naive, rather wrong thermodynamic model of a fluid, displayed in the temperature-pressure plane. Naively, we would expect the thermal expansion coefficient β to be near zero for the liquid phase and near $1/T$ for the vapor phase. (b) Contours of constant $T\beta$ for water, in the temperature-pressure plane. Note that β is substantially nonzero in the liquid phase, and is significantly greater than the ideal-gas value over most of the phase space shown.

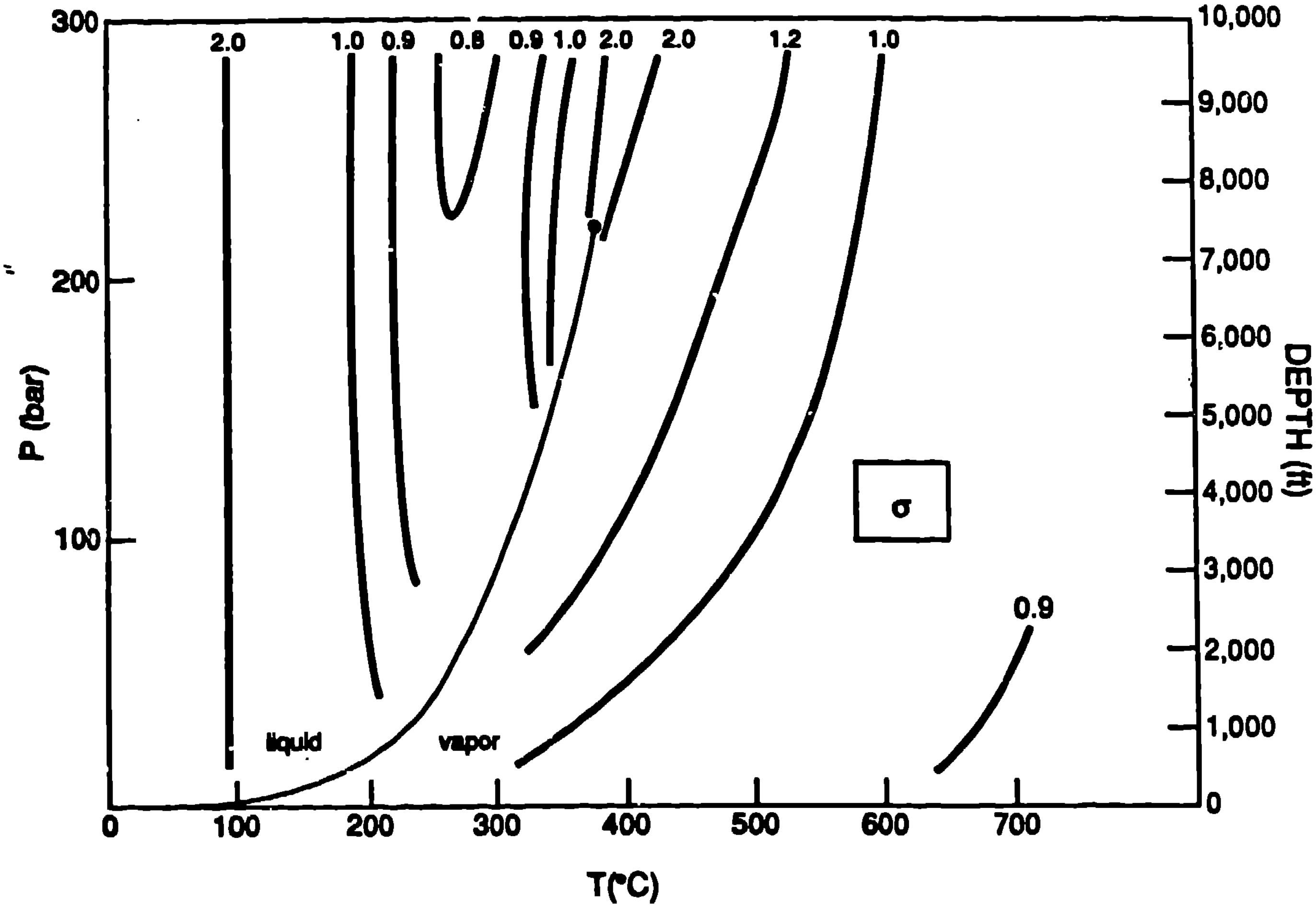

Fig. 2 Contours of constant Prandtl number σ for water, in the temperature-pressure plane.

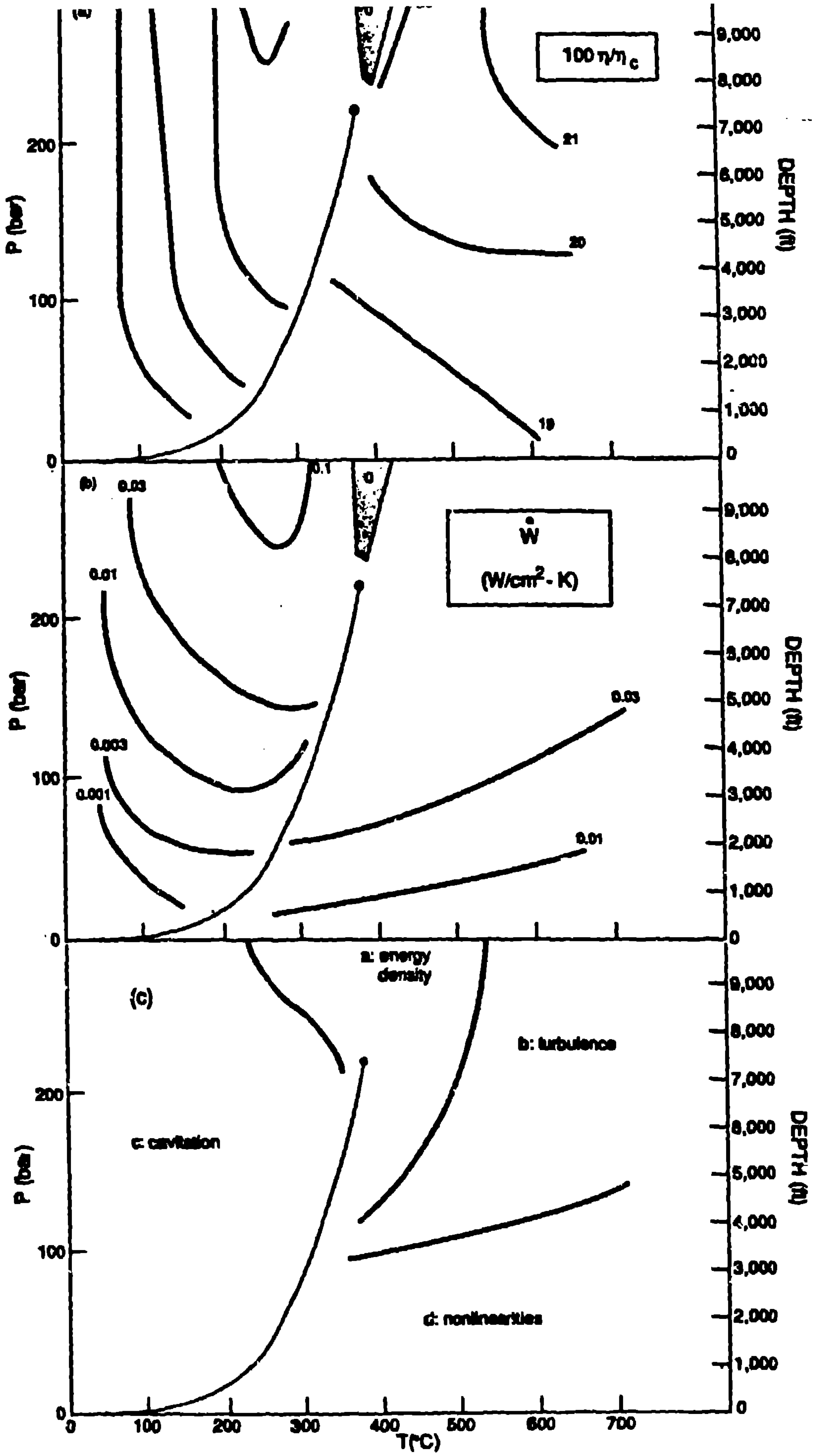

Fig. 3 Results of short-engine approximation for water at 100 Hz. (a) Contours of constant normalized efficiency. (b) Contours of constant normalized power density. (c) Map showing which of the four P_A-limiting effects listed in the text is operative.

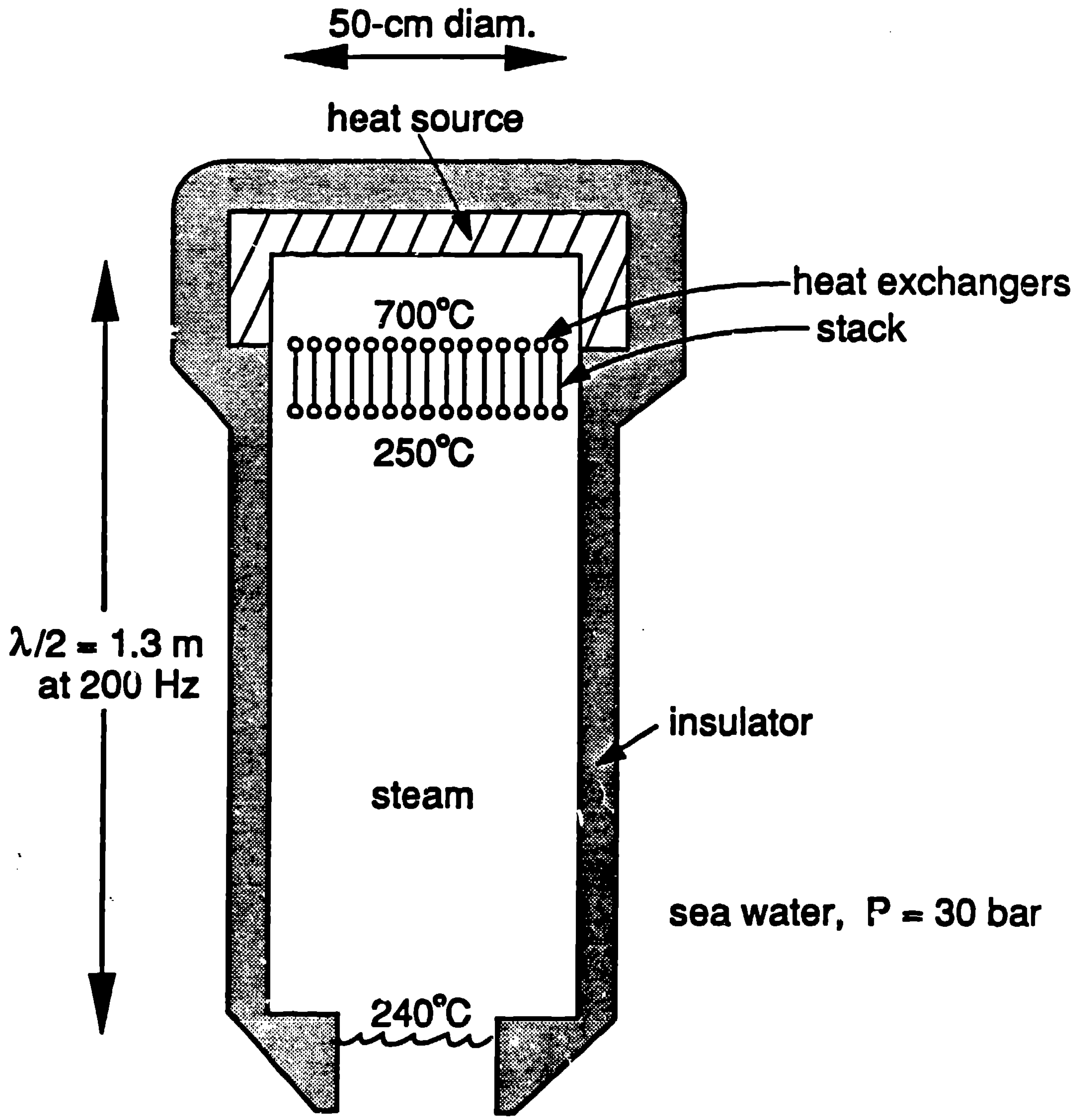

Fig. 4 Rough design of underwater thermoacoustic sound source.